



Advanced Control Surface Seal Development for Future Space Vehicles

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ABSTRACT

NASA's Glenn Research Center (GRC) has been developing advanced high temperature structural seals since the late 1980's and is currently developing seals for future space vehicles as part of the Next Generation Launch Technology (NGLT) program. This includes control surface seals that seal the edges and hinge lines of movable flaps and elevons on future reentry vehicles. In these applications, the seals must operate at temperatures above 2000 °F in an oxidizing environment, limit hot gas leakage to protect underlying structures, endure high temperature scrubbing against rough surfaces, and remain flexible and resilient enough to stay in contact with sealing surfaces for multiple heating and loading cycles. For this study, three seal designs were compared against the baseline spring tube seal through a series of compression tests at room temperature and 2000 °F and flow tests at room temperature. In addition, canted coil springs were tested as preloaders behind the seals at room temperature to assess their potential for improving resiliency. Addition of these preloader elements resulted in significant increases in resiliency compared to the seals by themselves and surpassed the performance of the baseline seal at room temperature. Flow tests demonstrated that the seal candidates with engineered cores had lower leakage rates than the baseline spring tube design. However, when the seals were placed on the preloader elements, the flow rates were higher as the seals were not compressed as much and therefore were not able to fill the groove as well. High temperature tests were also conducted to assess the compatibility of seal fabrics against ceramic matrix composite (CMC) panels anticipated for use in next generation launch vehicles. These evaluations demonstrated potential bonding issues between the Nextel fabrics and CMC candidates.

INTRODUCTION

CHALLENGES AND OBJECTIVES FOR CONTROL SURFACE SEAL DEVELOPMENT

High temperature structural seals have been identified as a critical technology in the development of future space vehicles. The current Shuttle orbiters require seals for their elevons and body flaps. In these locations the depth of section is large enough that relatively low temperature (<1500 °F) seals can be recessed a distance away from the outer mold line at the end of a tortuous air path that helps to insulate the seals from high heating rates. Smaller reentry vehicles currently being developed (e.g. X-37, Crew Return Vehicle) have less space allocated for seals (Fig. 1). This pushes the seals closer to or at the edge of the outer mold line and increases their operating temperature. Furthermore, the Shuttle's highly insulating tile system keeps heat from being conducted to the seals, while new vehicles are embracing hot CMC structures.¹ A combination of heat conduction through these CMC structures, heat convection to the seals and an inability to radiate heat from seal gaps results in seal temperatures upwards of 2600 °F. An additional challenge is that seals on the Shuttle are typically replaced after eight missions whereas all components of future fully reusable vehicles are expected to operate without refurbishment at least ten times longer. These conditions increase the seal design challenge.

Other previous studies demonstrated the need for control surface seals capable of operating at temperatures greater than 2000 °F. Rudder/fin seals on the X-38 vehicle were expected to reach 1900 to 2100 °F.² The current state-of-the art (SOA) seal design is a seal that is used in several locations on the Space Shuttle orbiters including the main landing gear doors, the orbiter external tank umbilical door, and the payload bay door vents. It was also the baseline seal design for the rudder/fin location of the X-38 vehicle (Fig. 1).² This seal has a nominal diameter of about 0.62 in. and consists of an Inconel X-750 spring tube stuffed with Saffil batting and overbraided with two layers of Nextel 312 ceramic sleeving (Fig. 2). Unfortunately these seal designs lose their resiliency and take on a large permanent set when they are compressed at high temperatures (Fig. 2). Permanent set limits the ability of a seal to conform to movements of the opposing sealing surface caused by structural and thermal loads and increases the chances of hot gas flow past the seal. Current control surface seals can also become damaged as they are

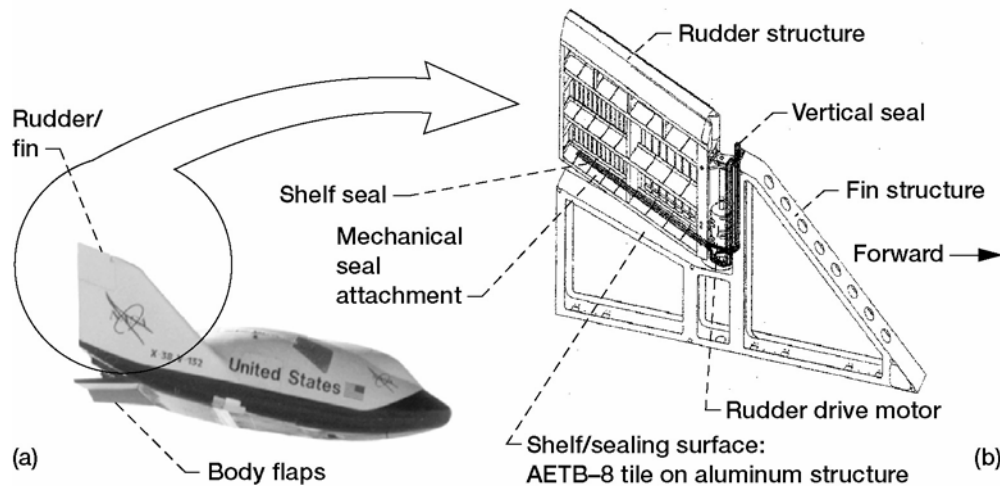


Figure 1.—Schematic showing (a) X-38 Crew Return Vehicle and (b) rudder/fin structure and seal locations.

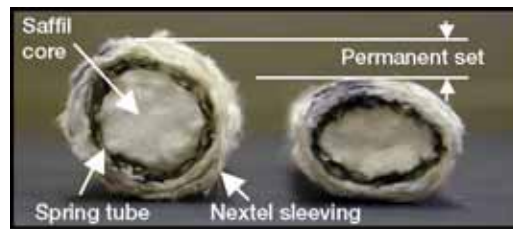


Figure 2.—Current baseline control surface seal before (left) and after (right) 1900 °F exposure showing loss of seal resiliency. Initial seal diameter is about 0.62 in.

scrubbed over rough sealing surfaces during actuation of the control surface.² If damage becomes severe, the amount of flow passing through the seals can increase to unacceptably high levels. Under these conditions, hot gas flow could reach underlying temperature-sensitive structures and damage them, leading to either degraded vehicle control or possible loss of vehicle and crew.

SEAL DESIGN REQUIREMENTS

Advanced seals have some very challenging design requirements. Table 1 summarizes the requirements for control surface seals on future reentry vehicles. Seals must survive at temperatures of up to 2600 °F while restricting hot gas leakage to underlying low-temperature structures to ensure vehicle structural integrity. Another important requirement is that the seals remain resilient enough after multiple temperature exposures to stay in contact with their opposing sealing surfaces. In some instances the sealing surface is a Shuttle-type thermal tile that cannot withstand high compressive loads. Therefore, the seals must maintain sealing contact with the surface without applying excessive loads to it. Thermal tiles and CMC's are also typically rather rough in their as-fabricated condition. As-fabricated surface roughnesses in the range of 515 to 574 μin . RMS are not uncommon, and if left unfinished these surfaces can cause unacceptable seal damage during actuation.³ As control surfaces are actuated the seals are swept over these rough sealing surfaces, and they must be wear resistant enough to withstand this scrubbing action without incurring excessive damage. GRC's goal is to design advanced control surface seals that are capable of meeting all of these requirements.

Table 1. Control surface seal design requirements

Design requirement	Goal
Seal temperature	Up to 2600 °F
Maximum unit loads	5 lbf/in. – Shuttle tile (Ref. 2) TBD – CMC
Pressure drop across seal	56 psf (Ref. 2)
Leakage	Minimize
Environmental considerations	Oxidizing environment
Use of cooling	Operate without active cooling
Reentry time	2200 sec (~37 min) (Ref. 2)
Size	0.5 to 1.0 in. nominal diameter
Reusability	TBD—nominally 10 to 100 cycles
Flexibility	Accommodate structural non-uniformities and seal around corners
Resiliency	Accommodate seal gap openings and closings
Seal gap size	Nominally 0.25 in.
Sliding speed	8 in./sec
Wear resistance	Withstand scrubbing against rough surfaces

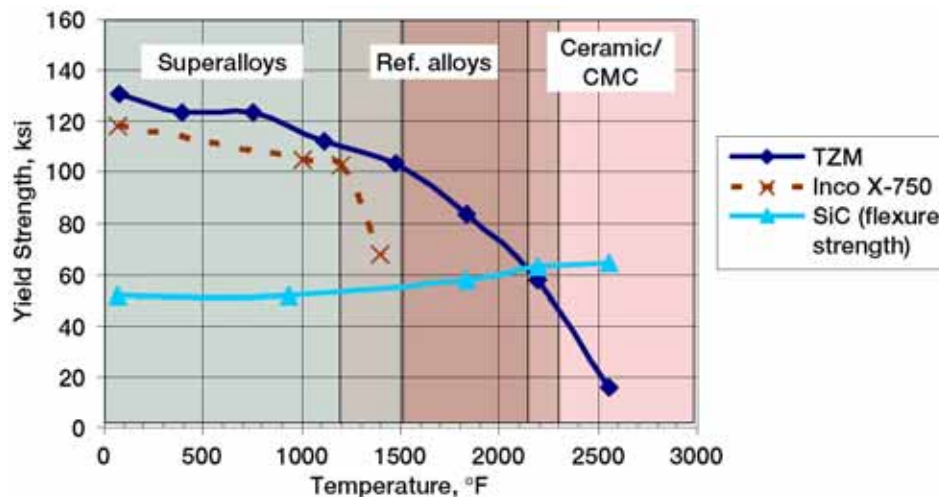


Figure 3.—Graph showing strength and approximate temperature usage ranges for several high temperature material systems.

SEAL DEVELOPMENT APPROACH: RESILIENCY IMPROVEMENTS

As stated previously, high temperature resiliency has been identified as a key shortcoming of the current baseline control surface seal. Therefore, much of the focus of the GRC Seals Team currently centers on enhancing the resiliency of next generation seals. These include improvements to the core and the spring element (Fig. 2) as well as use of a separate preloading device behind the seals. Changes to the core have been focused on engineering the core structure to impart additional resiliency. A previous investigation by the authors compared a seal with a packed core of uniaxial fibers versus a new design with a braided core of smaller seals.⁴ The seal tested in the current study utilized layers of smaller rope seals wrapped (or twisted) over one another. In addition to potentially improving seal resiliency, these seal designs can also provide enhanced core integrity during actual flight conditions. There have been instances on the Shuttle orbiter where portions of the unstructured seal core (i.e. Saffil batting) have been sucked out of the open end of a seal during flight, leaving a hollow seal that does not block the flow of hot gases as well. Operations staff have developed a hand-stitched fabric end cap to mitigate this problem, but this is a labor intensive process. The engineered core elements being investigated potentially offer a better solution to this issue due to the continuity of the core.

The primary problem with the internal spring element for the current baseline control surface seal is that the baseline material, a Nickel-based superalloy (Inconel X-750), dramatically loses strength above 1200 °F (with useable strength to about 1500 °F). As shown in Figure 3, one approach to resolve this issue is to substitute a higher-strength more creep resistant material. In order to extend the usable temperature range up to approximately 2300 °F,

a different class of materials, refractory alloys, must be considered. While these materials typically demonstrate superior high temperature strength and creep properties compared to most superalloys, they are generally very susceptible to oxidation. Consequently, these alloys must be coated with precious metals (e.g. Rh, Ir, Pt) or ceramic-based oxidation coatings. One promising refractory candidate, TZM (Mo-0.5Ti-0.08Zr) exhibits excellent strength properties at high temperature and is currently being evaluated as a candidate material for a spring preloader system. As shown in Figure 3, the only viable material class in terms of high temperature strength properties above 2300 °F is ceramics or CMC's. In contrast to the refractory alloys, these materials do not require oxidation coatings to survive in the harsh environments encountered during reentry operations. However, ceramic materials have limited elasticity making it difficult to fabricate seals or preloaders into complex shapes. The reduced elasticity also limits the "stroke" of these devices to accommodate large changes in gap size. Despite these challenges, ceramic-based preloaders have been fabricated for GRC and have shown promise in high temperature testing.⁴

The use of a separate preloader behind a thermal barrier seal offers a promising solution to the resiliency issue. These devices are being vigorously pursued at NASA GRC through internal investigations and outside contracted efforts.⁵ The incorporation of a discrete preloader behind a seal offers several potential benefits:

1. *Better resiliency*—In many applications the preloader will be insulated from the extreme heat by the thermal barrier seal and perhaps the surrounding groove material. This will facilitate retention of resiliency in the preloading device.
2. *Better control of force applied to opposing surfaces*—Many of the preloaders under investigation have geometries that can be easily altered to provide a desired stiffness and force range.
3. *Improved flow/heat blocking ability*—By using a separate preloader behind a thermal barrier seal, the properties of the combined sealing system can be optimized. For example, a softer preloader can be used to minimize the force applied to the opposing surface in conjunction with a denser seal to more effectively block high temperature air flow past the seal.

TEST PROCEDURES AND APPARATUS

SEAL SPECIMENS

Four seal designs were compared in this study. All of the seal designs were heat-cleaned using the 3M recommendations (1022 °F for 12 hrs.) to remove sizing prior to testing. Results for the baseline spring tube seal were obtained in previous investigations.² The other three seal designs incorporated engineered core modifications to possibly improve resiliency and reduce the potential for core extraction. The first two engineered core seals were evaluated in a recent investigation³ and the third design was tested for the current study. While these seals do not incorporate a resilient element and may be too stiff to be used by themselves in a control surface application, previous results have shown the use of a preloader behind these seals can significantly reduce the force applied to opposing structures.⁴

The first seal with an engineered core was a braided rope seal design originally developed by GRC during the NASP program.⁶ Nominally 0.600 in. in diameter, it consisted of a dense uniaxial core of 4000 yarns of 600 denier Nextel 312 fibers overbraided with two sheath layers of Nextel 550 fibers. This seal design will hereafter be referred to as the AC1 design.

To further improve the core structure and increase seal resiliency, GRC recently designed a seal with a core composed of smaller rope seals that were braided together (Fig. 4). The core of this seal was composed of three layers of 0.062 in. diameter rope seals in different configurations. The innermost layer of the core was made up of 7 of these seals in a uniaxial arrangement. Eight seals were braided over the inner layer to form the middle layer of the core, and then 16 seals were braided over the middle layer to form the outer layer of the core. Over this engineered core, two sheath layers were then braided to create a nominal overall diameter of about 0.565 in. This seal design was made entirely of Nextel 440 ceramic fibers and will hereafter be referred to as the BC1 design.

The third seal configuration tested (herein referred to as TC1) was an alternate approach to improving seal resiliency and flow blocking ability as compared to the BC1 design. As shown in Figure 5, the seal consisted of alternating layers of helical wrap seals and sheath layers twisted around a central core of smaller seals. Four sheath layers of Nextel fabric were used to enclose the seal and to achieve the final diameter of about 0.600 in. This seal design was also made entirely of Nextel 440 ceramic fibers.

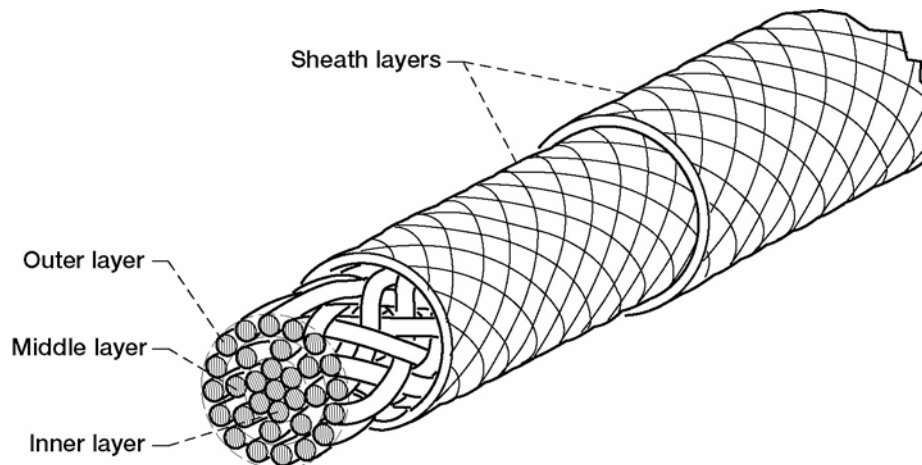


Figure 4.—Schematic of BC1 seal design showing sheath layers and inner, middle, and outer layers of core.

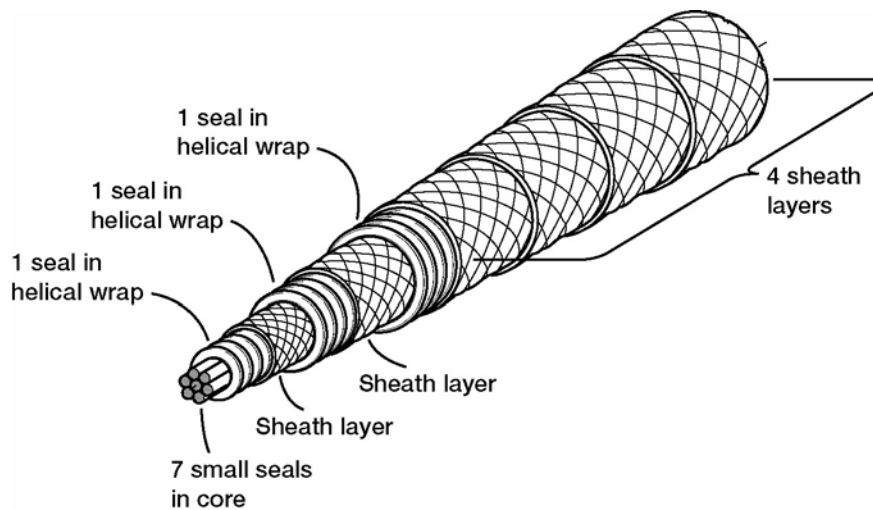


Figure 5.—Schematic of TC1 seal design showing sheath layers, alternating helical wrap and sheath layers, and inner core.

SEAL PRELOADING DEVICE SPECIMENS

Four inch samples of the Inco X-750 spring tube were tested to assess performance at several different temperatures and determine the temperature at which the resiliency significantly degrades. The spring tubes (Boeing Specification MB0160-047, ST5, 10N-3S-.009D-4.9 cpi) had a nominal tube outer diameter of 0.560 ± 0.025 in. and used three strands of 0.009 in. diameter wire. Ten needles (loops) were used along the circumference of the tube with a lengthwise loop spacing of 4.9 courses per inch (cpi).

A canted coil spring (CCS) produced by Bal Seal Engineering Company, Inc. (Fig. 6) was also evaluated as a seal preloading device for this study. These types of springs have several unique features that could make them very good preloading devices. Unlike typical compression springs that generate increasing amounts of force as they are compressed, the force produced by canted coil springs remains nearly constant over a large deflection range. This is an appealing feature for a seal preloading device because it could provide a large amount of stroke and resiliency to a seal without applying excessive loads to opposing sealing surfaces. Another advantageous feature of canted coil springs is that they are produced in long, linear lengths that would allow them to be installed in a groove directly behind a seal and potentially around corners. Additionally, the part count would be far lower for a canted coil spring than for a typical compression spring because hundreds of compression springs would have to be lined up behind a long seal to accomplish what only a few canted coil springs could do. The springs evaluated in this study

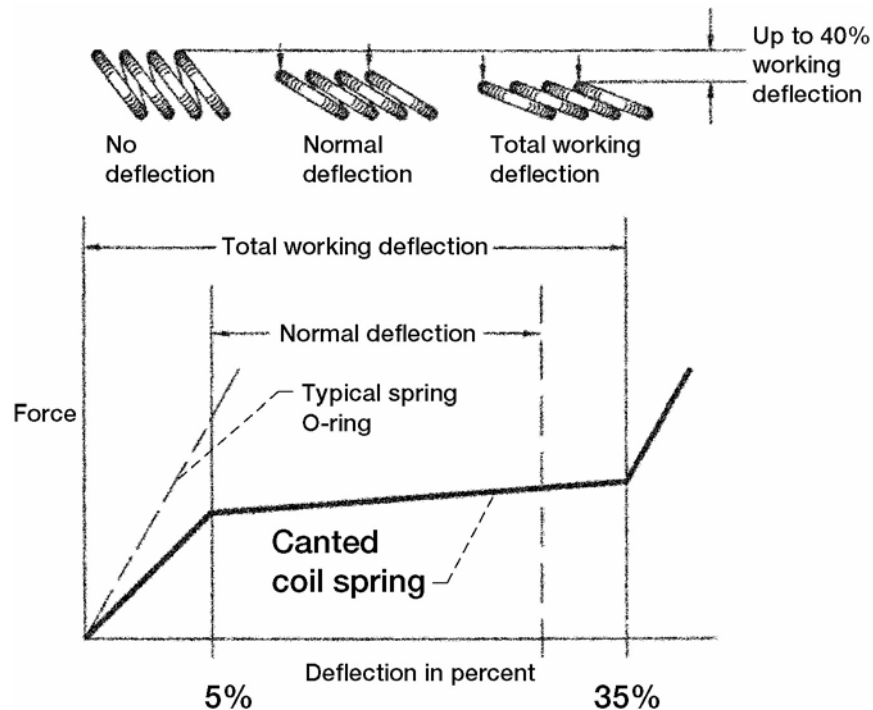


Figure 6.—Schematic of deflection response (bottom) of a typical canted coil spring (top).

were Bal Seal part number 109MB-(84)L-2 and were made of 302 stainless steel. They had a wire diameter of 0.041 in. and a coil height and width of 0.450 in. and 0.508 in., respectively. The authors recognize that stainless steel springs would not be suitable for 2000+ °F service and that other material systems would have to be used for these higher temperatures.

COMPRESSION TESTS

Test Apparatus

A series of room temperature and high temperature (up to 2000 °F) compression tests were performed on the seals and preloading devices using a new SOA test rig that was recently installed at GRC. This test rig is capable of performing either high temperature seal compression tests or scrub tests at temperatures of up to 3000 °F by using different combinations of test fixtures made of monolithic silicon carbide (Hexoloy α -SiC). The main components of this test rig are a servohydraulic load frame, an air furnace capable of 3000 °F, and a non-contact laser extensometer (Fig. 7). The 500 lb load cell used in this study had an accuracy of ± 0.15 lb (± 0.03 percent of full scale), and the accuracy of the laser was ± 0.00025 in. Further details of this rig can be found in the paper by Dunlap, et al.⁴

Compression tests were performed inside the furnace using the test set up shown in Figure 7. These tests were performed to determine seal resiliency and stiffness and to generate seal load versus displacement (i.e. linear compression) data at room temperature and high temperatures. Test specimens were installed into a seal holder that rested on a stationary base at the bottom of the furnace. The samples tested were nominally 4 in. long. A movable platen was attached to a loading rod that passed through the top of the furnace and connected to the water-cooled coupling above the furnace. This platen was actuated up and down to load and unload test specimens. The laser extensometer was used to measure the amount of linear compression that the test specimens were under during testing. Linear compression was also monitored using an external extensometer with a ± 1.0 in. range (± 0.0058 in.).

Test Procedure

Table 2 summarizes the testing parameters used to evaluate the candidate samples. At the beginning of each test, initial contact between the test specimen and the load platen was defined when the load on the specimen reached the initial preload presented in the table. The samples were then compressed to a given compression level at a particular rate, held for a certain dwell period, and then fully unloaded at the specified rate. This cycle was then

repeated the prescribed number of times. The Inco X-750 spring tubes were evaluated using this procedure at room temperature, 1200, 1500, 1750, and 2000 °F. The canted coil spring was tested only at room temperature and the seals (AC1, BC1, and TC1) were tested at room temperature and 2000 °F. Primary and repeat tests were performed for each test case, and a new specimen was used for each test.

For the room temperature tests performed on the seals, a pressure-sensitive film was placed in between the seal specimens and the movable platen for the first load cycle to determine the contact width and length of the specimen as it was compressively loaded. The film was removed after the first load cycle, and the seal footprint length and width were then used to calculate seal preload in psi. The film could not be used for the tests performed at 2000 °F.

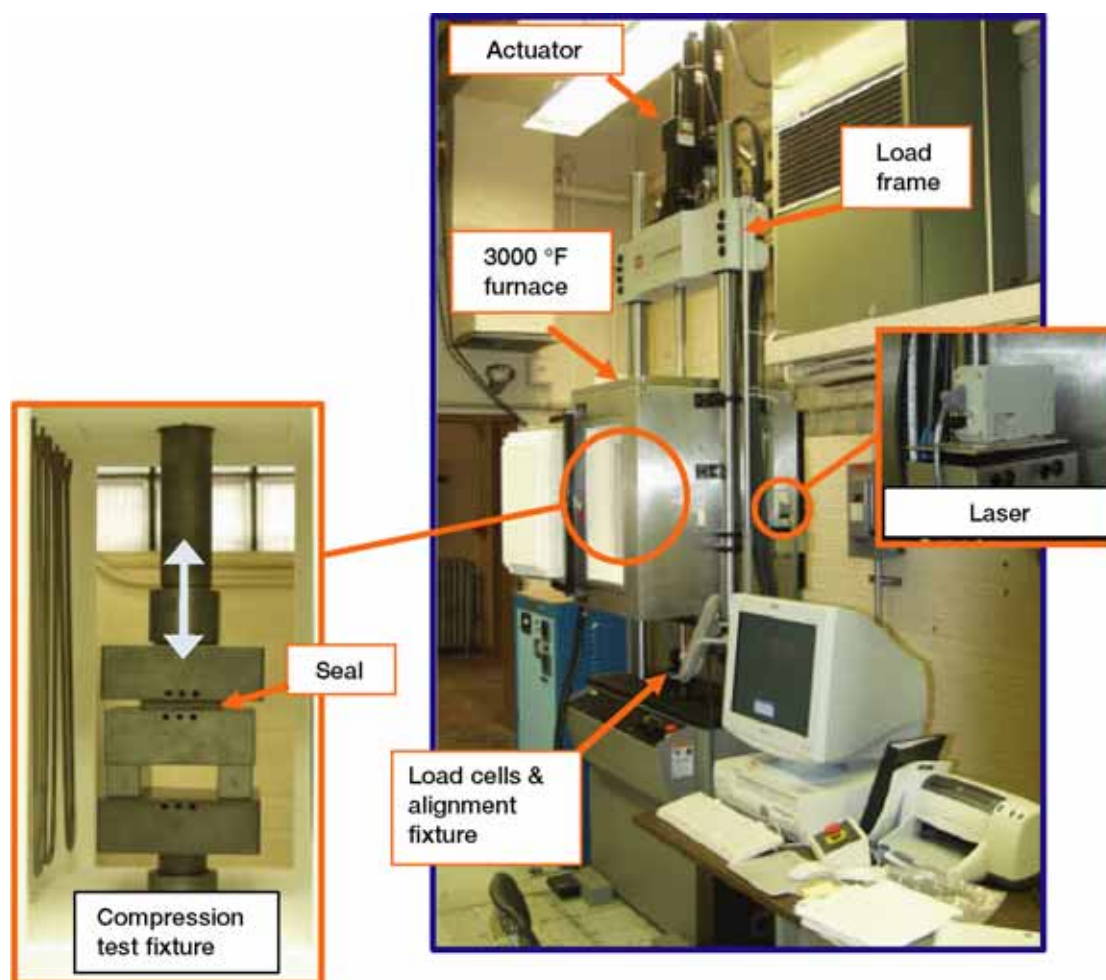


Figure 7.—Photograph of hot compression and scrub test rig showing main components: load frame, high temperature furnace, laser extensometer, and high temperature compression fixturing.

Table 2. Summary of testing parameters for compression tests

	Initial preload, lbf	Compression level, in.	Cycles	Dwell, sec	Load rate, in./sec	Unload rate, in./sec
Spring tube	0.2	0.112	20	250	0.002	0.002
AC1 (at RT and 2000 °F)	1.0	0.120	20	60	0.001	0.001
BC1 (at RT and 2000 °F)	1.0	0.113	20	60	0.001	0.001
TC1 (at RT and 2000 °F)	1.0	0.120	20	60	0.001	0.001
AC1 + CCS	1.0	0.120	20	60	0.001	0.001
BC1 + CCS	1.0	0.113	20	60	0.001	0.001
TC1 + CCS	1.0	0.120	20	60	0.001	0.001

FLOW TESTS

Test Apparatus

Room temperature flow tests were performed in a linear flow fixture shown in Figure 8. The flow fixture was designed so that seals of different diameters could be tested in removable cartridges that are inserted into the main body of the test fixture. Seals can be tested in this fixture with different seal gaps and under different amounts of linear compression. Flow meters upstream of the flow fixture measured the amount of flow that passed through the test seal. The maximum capacity flow meter that was used had a range of 0 to 26 standard cubic feet per minute (SCFM) with an accuracy of 1 percent of full scale. A pressure transducer (0 to 5 psid, accuracy 0.051 percent of full scale) upstream of the test seal measured the differential pressure across the seal with respect to ambient conditions, and a thermocouple measured the upstream temperature.

Test seals were mounted in the groove of a seal holder that was inserted into the test cartridge (Figs. 8b and 8c). The groove was 0.62 in. wide and 4 in. long. The amount of preload, or linear compression, applied to the seals was varied by placing metal shims in the groove behind the seal. The test fixture was originally set up to test 12-in.-long seals. To test the shorter 4 in. seals, aluminum filler blocks with O-ring grooves in them were installed on either side of the seal assembly to seal the outboard seal ends (Fig. 8c). Further details of this test rig can be found in the paper by Dunlap, et al.⁴

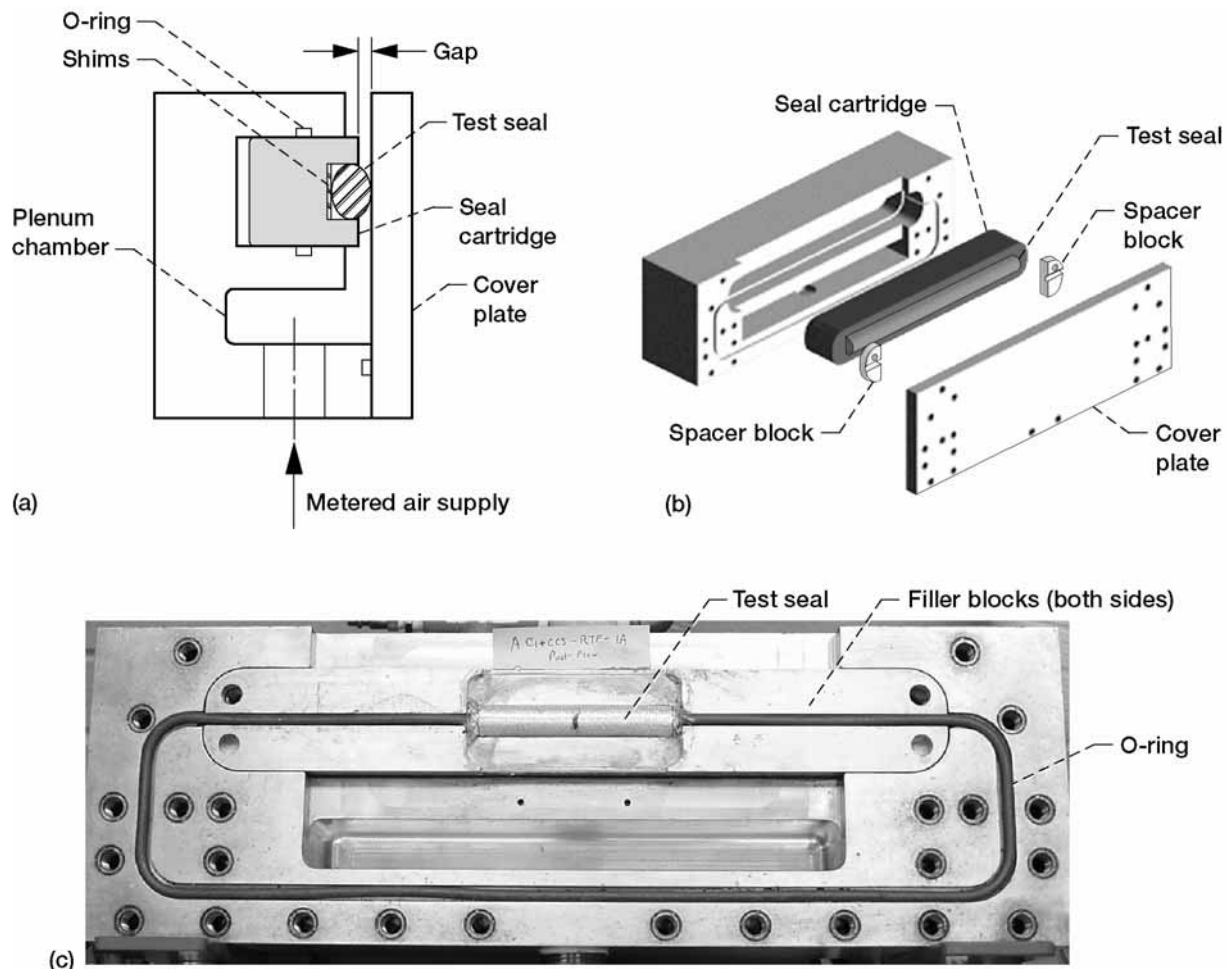


Figure 8.—Schematic of flow fixture. (a) Cross section. (b) Isometric. (c) Front view of 4 in. seal specimen installed in flow fixture showing grooved filler blocks that join O-ring to both ends of seal.

Test Procedure

Room temperature flow tests were performed on the seal designs at a nominal compression level of 20 percent of the specimen's overall diameter. These tests were conducted on as-received seals and were also performed on seal designs with canted coil springs installed behind them to determine how this affected the flow blocking ability of each design. For these evaluations, deeper grooves were used to account for the height of the springs. A new seal specimen was used for each flow test. All flow tests were performed using a 0.625 in. wide groove and a 0.250 in. seal gap.

COMPATIBILITY TESTS FOR CMC CONTROL SURFACE MATERIALS AGAINST SEAL FABRICS

Test Apparatus

A series of compatibility tests were performed between two candidate CMC control surface materials (carbon fiber reinforced silicon carbide, C/SiC, and carbon fiber reinforced carbon, C/C) and two Nextel fabrics (Nextel 440 and Nextel 720) to assess bonding propensity. These fabric materials are being considered for use in the outer sheath layers of future control surface seals. Tests were conducted using a high temperature air furnace capable of approximately 2800 °F. The furnace was equipped with top and bottom loading ports and SiC rods to apply simulated pressure loads to the CMC-Nextel pairs. In addition, the CMC test panels were mated against SiC test plugs to assess bonding propensity for future hot compression and hot scrub tests (Fig. 9).

Test Procedure

The C/SiC material was cut into approximately 1 in. x 1.25 in. samples for testing. A large panel of the C/C CMC was coated with C-CAT SiC/TEOS and Type A sealant on the top and bottom surfaces and then cut into 1 in. by 1 in. samples for testing. After cutting the samples, the edges were coated with Ceraset (Dupont) coating. Figure 9 shows a typical test setup for these tests. Samples of the CMC candidates were sandwiched between SiC test fixturing and specimens of the fabric materials previously mentioned. Prior to testing, all of the Nextel fabrics were heat-cleaned using the 3M recommendations (1292 °F for 5 min.) to remove sizing. The stack was loaded using a 5 lb weight placed on top of the SiC loading rod which resulted in contact pressures of approximately 7-8 psi. After loading the specimens, the furnace was heated to roughly 2650 °F at 500 °F/hr. Under these conditions, the samples spent approximately 1.5 hrs at temperatures of 2600+ °F.

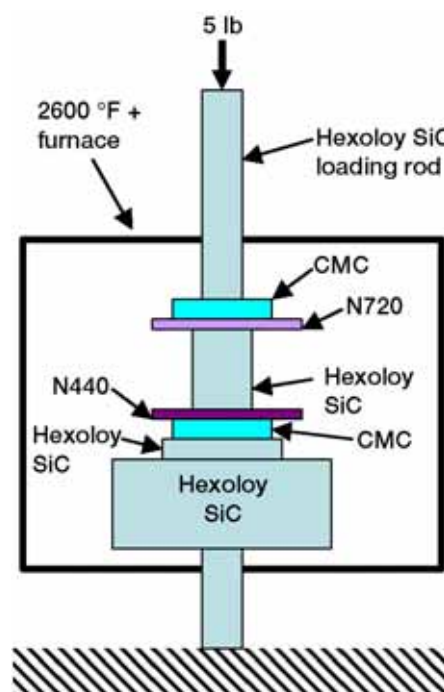


Figure 9.—Schematic of typical test setup for compatibility tests between seal fabrics and CMC control surface materials.

RESULTS AND DISCUSSION

COMPRESSION TEST RESULTS: SPRING TUBES AT HIGH TEMPERATURE

A plot of percent residual interference versus test temperature for the Inco X-750 spring tube at the beginning of cycles 2 and 20 is presented in Figure 10. As shown by the graph, a significant drop in percent residual interference (resiliency) occurred between 1200 and 1500 °F. The percent residual interference was defined as:

$$\% \text{ Residual interference} = (\text{Total linear compression} - \text{Permanent set}) / \text{Total linear compression} * 100$$

The amount of permanent set for subsequent load cycles was identified as the displacement when the load reached the defined initial preload value.

The significant drop in residual interference after 20 load cycles was visually confirmed in the specimens after testing, as shown in Figure 11. Not surprisingly, this drop in performance mirrored the temperature dependent yield strength behavior of this alloy. The plot also demonstrates that repeated cycling further reduced the resiliency of the spring tube element. Guidelines for maximum use temperatures to retain a specified percent resiliency can also

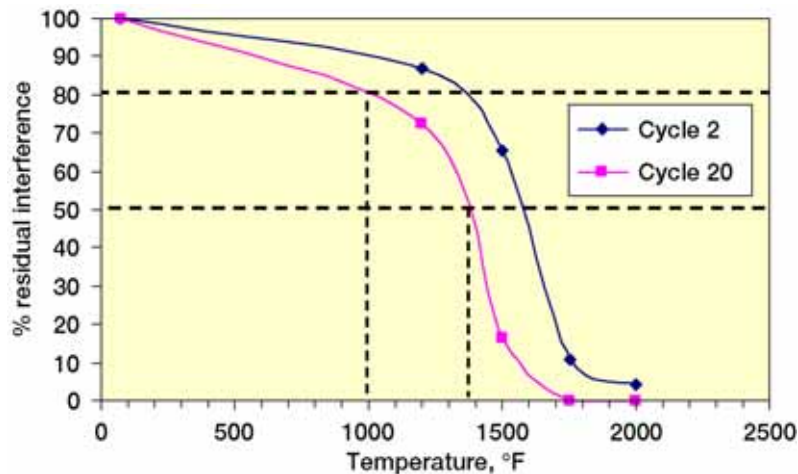


Figure 10.—Plot showing percent residual interference versus test temperature for the baseline Inco X-750 spring tube.

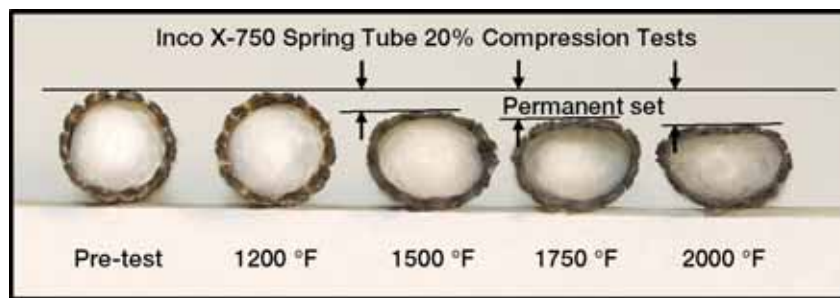


Figure 11.—Picture showing permanent set of Inco X-750 spring tubes used in baseline spring tube seal design at 20% compression and various high temperatures. (Note: Cotton stuffed into spring for photograph).

be obtained from the graph. As an example, for 20 cycles, the maximum use temperature for 80 percent springback was approximately 1000 °F. For 50 percent residual interference, the use temperature increased to about 1400 °F. At 2000 °F, the spring tube exhibited no discernable resiliency after 20 load cycles. This is a concern for future seal applications required to operate at temperatures above 2000 °F.

COMPRESSION TEST RESULTS: ADVANCED CONTROL SURFACE SEALS AND CANTED COIL SPRING PRELOADERS

Figure 12 shows a comparison of percent residual interference values for the baseline control surface seal and different iterations of the engineered core design. Previous tests conducted on the baseline spring tube seal using a slightly different test setup showed that at the beginning of the fourth compression cycle, the percent residual interference was approximately 68 percent.² Similar tests on this seal conducted after a 1900 °F exposure while under compression resulted in a significant drop in percent residual interference to 15 percent.

As demonstrated by Figure 12, modifications to the core alone did not appear to significantly improve the resiliency at either room temperature or high temperature. At room temperature, the baseline seal design was considerably better than the engineered core alternatives. This was not surprising since this seal had an integral spring element while the engineered core designs did not. However, when a spring preloader was placed behind these new seal candidates, the resiliency significantly increased surpassing that of the baseline design. A 26 percent improvement in room temperature resiliency was observed with the TC1 design on top of the canted coil spring.

At high temperatures (e.g. 1900 °F), the baseline design suffered a significant loss in resiliency. This result was expected because the spring tube loses most of its resiliency at temperatures greater than 1500 °F, as discussed in the previous section. A high temperature preloader behind the thermal barrier seal would possibly be exposed to a much lower temperature due to the thermal barrier and therefore would have better resiliency than an internal spring element.

The use of a spring preloader can also significantly reduce the force applied by the seal to the opposing structure, as shown in Figures 13 and 14. Tests performed on the TC1 design without a spring produced a peak load of 35.2 lbf/in., while addition of a “medium” canted coil spring behind the seal reduced the peak load to 6.2 lbf/in. As shown in Figure 13, this was much closer to the load level of the baseline spring tube seal. The employment of a less stiff spring element could result in an even closer match to the performance of the baseline seal. This becomes critical when the seal is mated against delicate surfaces like Shuttle tiles.

As opposed to the spring tube seal, the seal with the canted coil preloader had the added benefit of exhibiting a region toward the end of the stroke where the load began to level off (Fig. 13). This means the seal/spring system can accommodate relatively large variations in gap due to thermal growth or other movements without applying excessive force to surrounding structures.

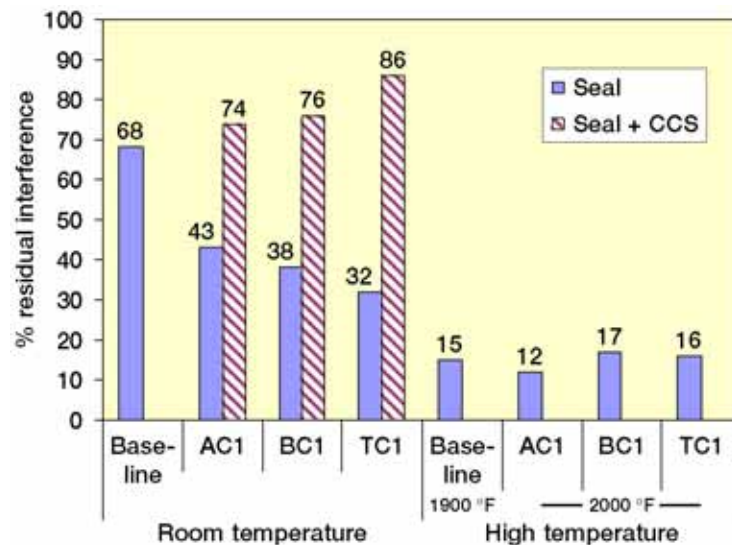


Figure 12.—Plot showing effect of temperature, core structure, and use of canted coil spring preloader on percent residual interference for the fourth load cycle at 20% compression.

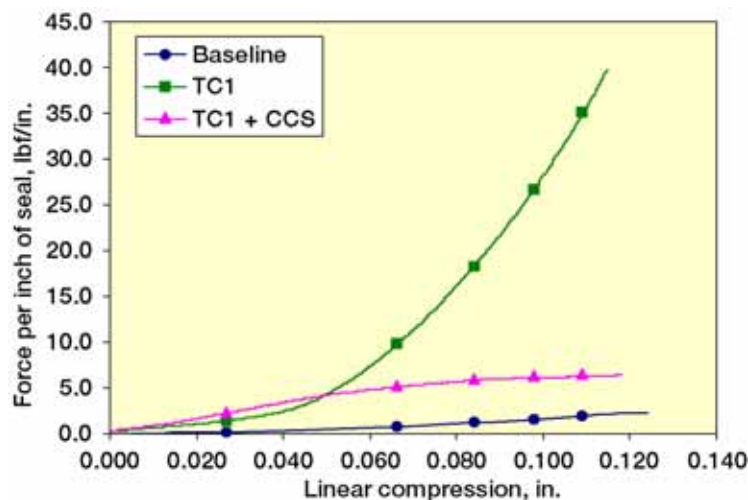


Figure 13.—Plot comparing force versus displacement during cycle 1 for the baseline spring tube seal versus the TC1 design with and without the canted coil spring preloader. (Note: Symbols shown for clarity; do not represent actual data points).

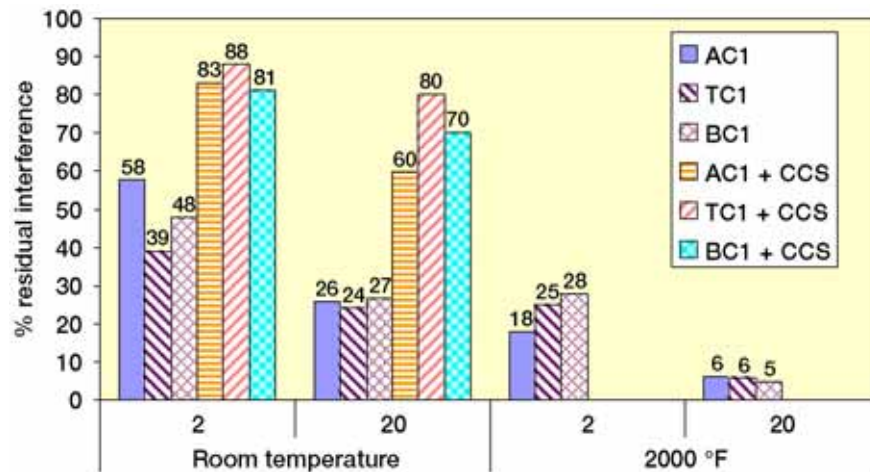


Figure 14.—Plot showing effect of temperature, load cycling, and presence of canted coil spring on percent residual interference for nominal 20% seal compression. (Values shown are average of two or more tests).

Table 3. Summary of seal peak loads and residual interference as a function of temperature and presence of canted coil spring

Cycle	Peak load at dwell, lbf/in.					Residual interference at start of load cycle, in.					Percent residual interference at start of load cycle, %				
	1	2	3	10	20	1	2	3	10	20	1	2	3	10	20
AC1 at room temp.	56	38	31	19	15	0.117	0.068	0.057	0.037	0.030	100	58	49	32	26
BC1 at room temp.	24	21	19	15	14	0.111	0.053	0.046	0.034	0.029	100	48	41	31	27
TC1 at room temp.	35	31	28	23	21	0.117	0.046	0.041	0.032	0.029	100	39	34	27	24
AC1 at 2000 °F	46	36	30	19	11	0.117	0.022	0.017	0.009	0.007	100	18	14	8	6
BC1 at 2000 °F	23	15	13	8	5	0.112	0.031	0.024	0.010	0.005	100	28	21	9	5
TC1 at 2000 °F	82	59	49	29	25	0.120	0.030	0.022	0.010	0.007	100	25	18	9	6
AC1 + canted coil spring at room temp.	5	5	5	5	5	0.120	0.099	0.092	0.078	0.071	100	83	77	65	60
BC1 + canted coil spring at room temp.	5	5	5	5	5	0.112	0.091	0.088	0.082	0.079	100	81	78	73	70
TC1 + canted coil spring at room temp.	6	6	6	6	7	0.121	0.106	0.104	0.100	0.096	100	88	87	83	80

Compression test results for the AC1, BC1, and TC1 seal configurations are further compared in Table 3. The peak load during the dwell segment for cycles 1, 2, 3, 10, and 20 are presented in the table as well as the residual interference values for the corresponding cycles. The percent residual interference for the same cycles is also shown in this table. These results were the average of two tests, with the exceptions of BC1 at room temperature and TC1 at high temperature. For these cases, three results were averaged.

The percent residual interference results for the engineered core designs are also plotted in Figure 14. As depicted in the graph, the resiliency of the seals decreased with load cycling. For the tests conducted at room temperature on seals without preloaders, the AC1 design appeared to possess the best resiliency. By the 20th cycle at both room temperature and high temperature no significant difference in the residual interference values could be observed. The graph also shows a notable decrease in residual interference as a function of temperature for all the seal candidates. A visual inspection of the seal candidates after high temperature testing revealed that the seals were noticeably flatter on top when compared to the pre-test condition. Perhaps the most encouraging aspect of this data was the considerable increase in room temperature resiliency generated when the seals were used in conjunction with the canted coil spring preloader. By the 20th load cycle, the seals with canted coil springs behind them showed a 2.3 to 3.3x improvement in resiliency when compared to the seals alone. The TC1+CCS design also appeared to sustain the most improvement as the test specimens were cycled.

Figure 15 presents a typical plot of unit force versus linear compression for the TC1 design as a function of temperature and the inclusion of a canted coil spring preloader. As shown in this figure, the room temperature test and 2000 °F test of the seal alone showed a large drop in load capacity with cycling. At room temperature the drop in load between the 1st and 20th cycles was 15 percent; at 2000 °F the decrease was 72 percent. By contrast the TC1 seal on top of the canted coil spring preloader showed much better load retention.

The plot also depicts a significant increase in peak load at 2000 °F compared to the ambient temperature results. Although the exact reasons for this increase are unknown, the phenomenon appeared to be real and was consistently observed during three separate high temperature tests. Further testing will be required to resolve this. However, by the 20th load cycle the loads at room temperature and high temperature were comparable.

A comparison of the peak loads for cycles 1 and 20 for the engineered core designs is shown in Figure 16. As expected, the AC1 design exhibited the highest peak load during the first cycle at room temperature. This core was more densely packed compared to the other core designs, so a stiffer seal would be anticipated. A measure of the contact width for this seal using the pressure-sensitive film previously described showed that the AC1 had a contact width of 0.362 in. and yielded a unit load of 155 psi for the first cycle. By comparison, the TC1 design had a lower contact width (0.242 in.), but generated a similar unit load of 145 psi. The BC1 design produced a width of 0.307 in. and a corresponding load of 81 psi.

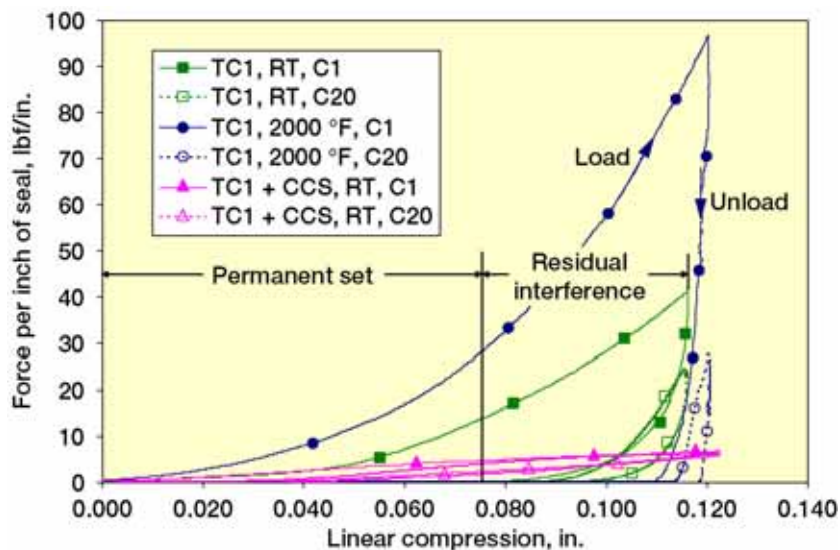


Figure 15.—Plot showing effect of temperature and presence of canted coil spring on unit load versus displacement for TC1 seal design during cycles 1 (C1) and 20 (C20). (Note: Symbols shown for clarity; do not represent actual data points).

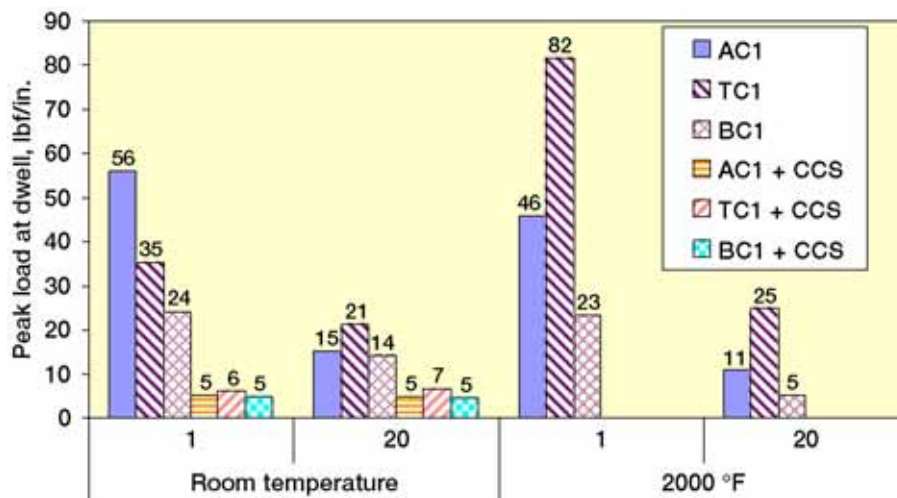


Figure 16.—Plot showing effect of temperature, load cycling, and presence of canted coil spring on peak loads for nominal 20% seal compression. (Values shown are average of two or more tests).

After 20 load cycles at room temperature, however, the AC1 design showed similar loads to the other seals as it was compacted in the seal groove. Both the TC1 and BC1 designs also demonstrated decreases in load capacity after 20 cycles, but the changes were not as severe as with the AC1 seal.

At 2000 °F for cycle 1, the TC1 design yielded the highest peak load with an increase of over 2x compared to its room temperature load. As stated earlier, the reasons for this significant increase in peak load for the TC1 seal are unclear. By contrast, the AC1 and BC1 designs softened slightly at high temperature. However, by the 20th load cycle the effect of temperature on the TC1 seal was not as obvious when the room temperature (21 lbf/in.) and the 2000 °F (25 lbf/in.) were compared. The AC1 and BC1 designs continued to exhibit a decrease in load at high temperatures with the BC1 design showing the largest drop.

The seals on top of the canted coil springs demonstrated similar peak loads for all the candidates with no discernable degradation in performance as a function of cycling. These loads were also comparable to those recorded for the canted coil springs by themselves.⁴ These results along with the resiliency improvements documented previously illustrate the potential and importance of development of a high temperature version of this type of preloader device.

FLOW TEST RESULTS

A comparison of flow results for the seal designs is presented in Figure 17. As illustrated by the graph, both the AC1 and TC1 engineered core designs exhibited lower flow than the baseline spring tube seal. The BC1 design demonstrated somewhat higher leakage rates than the baseline seal, but this was likely due to the seal being undersized relative to the groove by approximately 0.060 in. The AC1 design showed the lowest flow with leakage rates about 1/6 that of the baseline design at 144 psf (1.0 psid). The results for the AC1 and TC1 designs are not surprising as the baseline seal was probably more porous than the modified core designs. Previous investigations on the spring tube design showed that the porosity of this seal was approximately 85%.² Porosity testing on the engineered core designs was not conducted for the current investigation, but one could reasonably infer from that load data (e.g. stiffness correlates with density) that these designs were more dense than the baseline. Future testing will confirm this hypothesis.

Due to their improved flow-blocking capabilities, the engineered core seals themselves might be suitable in CMC applications where designers are still establishing upper load limits. However, they could not be used in applications using Shuttle tile where unit loads are limited to approximately 5 lbf/in. The use of a less stiff preload device behind the seal would be necessary and therefore this type of setup was also tested. When used with canted coil springs, all of the engineered core designs showed an increase in flow rate relative to the seals by themselves. For these cases, only the AC1 seal with the canted coil spring remained below the baseline flow values. This increase in leakage was likely due to the fact that the seals were undersized relative to the groove size (with the BC1 design being the most undersized). With no spring installed behind the seals, they were able to deform and contact the side walls of the groove when they were compressed. This configuration allowed the seals to fill the groove and restrict flow better. For the tests in which a spring was installed behind the seals, much of the deformation likely occurred in the spring causing less contact between the undersized seals and the groove side walls. Careful sizing of the seal relative to the groove size should minimize this issue.

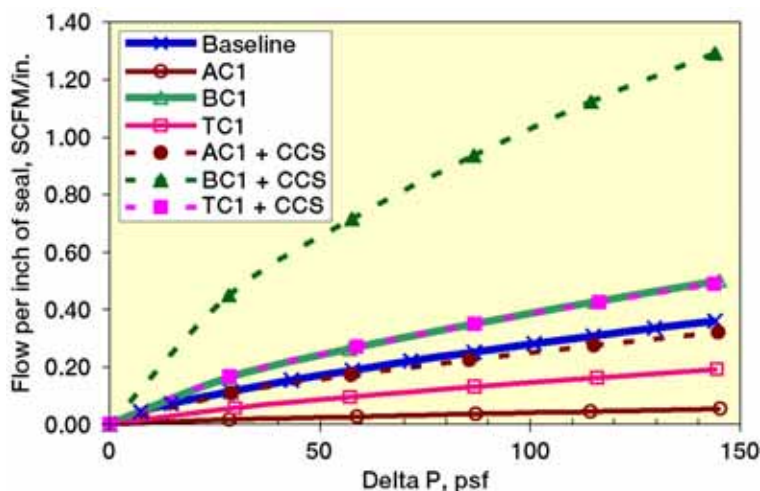


Figure 17.—Plot comparing flow results for baseline spring tube seal versus select engineered core designs with and without a canted coil spring preloader.

COMPATIBILITY TESTS RESULTS FOR CMC CONTROL SURFACE MATERIALS AGAINST SEAL FABRICS

Representative photographs of the results from the CMC compatibility tests are shown in Figure 18. Results with the C/SiC samples demonstrated varying degrees of stickage to the Nextel fabric materials. In the worst case, portions of the Nextel material were ripped off when removal of the fabric from the CMC was attempted, as shown in Figure 18a. The C/SiC showed minimal stickage to the Hexoloy material in all test cases. It was also noted that the Nextel 720 material appeared to be more brittle than the Nextel 440 candidate after the long term heat exposure in air.

Typical results for the C/C CMC test panels are shown in Figure 18b. This material exhibited more severe stickage to the Nextel fabrics and the CMC samples. In addition, the C/C specimens stuck to the Hexoloy fixturing samples. This bonding of materials is likely due to the coatings used to protect the C/C material, though further evaluations are required.

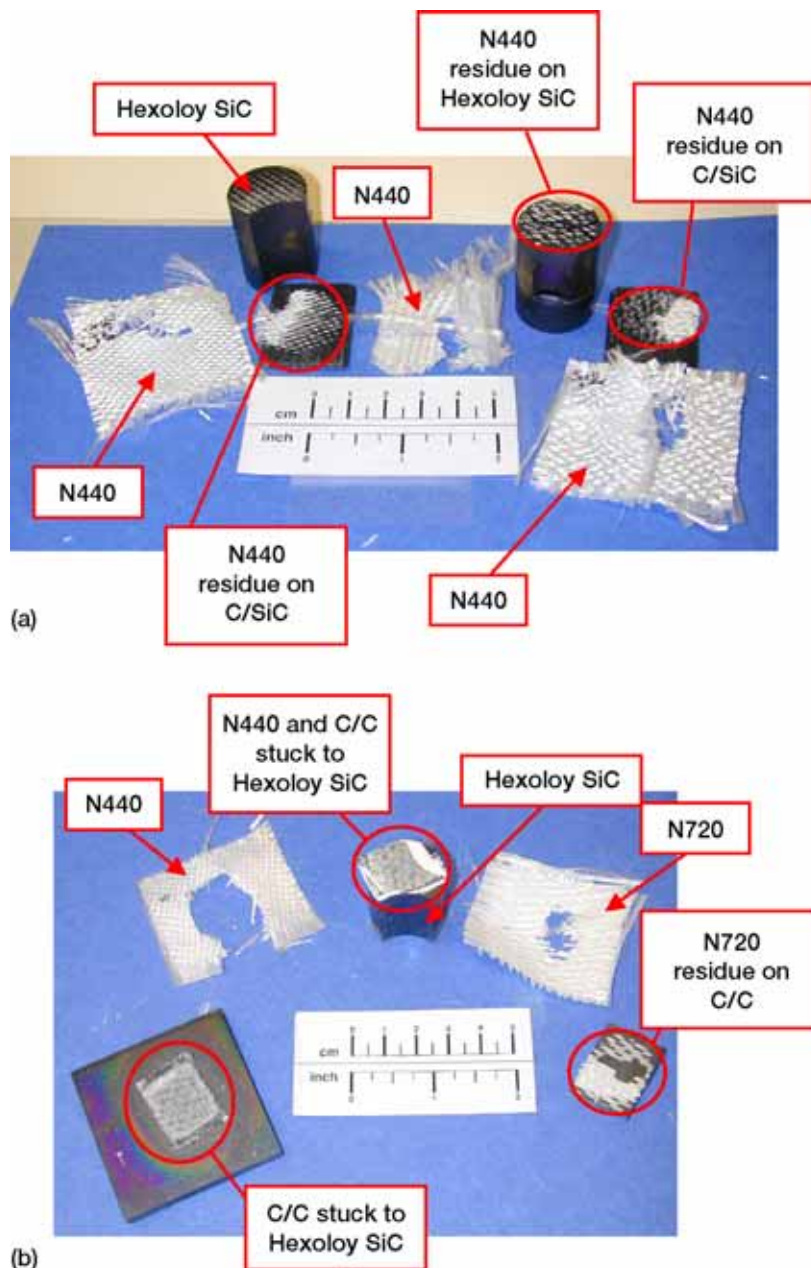


Figure 18.—Photographs of specimens used in select compatibility tests for (a) C/SiC and (b) C/C test panels.

It was also observed that the C/C CMC material significantly degraded (oxidized) after exposure at 2600 °F in air for the extended test period. Failure of the hand-applied Ceraset coating is believed to be the reason for the disintegration of these test panels. This “temporary” coating would likely not be used in a flight application. It should also be pointed out that for all tests, the samples experienced the transients of furnace heatup and cooldown and thus were exposed to high temperatures for many hours. In the actual application, the exposure times would probably be significantly shorter (in terms of minutes as opposed to hours).

SUMMARY AND CONCLUSIONS

The current baseline control surface seal does not meet the demanding requirements for advanced space vehicles. This seal exhibits poor resiliency at high temperature which would likely permit excessive leakage of high temperature gases past the seal into temperature sensitive structures. In addition, the hand stuffed Saffil batting has been extracted on occasion during Shuttle flights. Due to these deficiencies, NASA GRC developed and conducted testing on the several new control surface seal candidates. Based on room temperature and 2000 °F compression testing as well as flow testing, the following conclusions were noted:

1. The baseline Inco X-750 spring tube showed a significant drop in resiliency at temperatures above 1200 °F. After 20 cycles of high temperature compression testing, the spring tube retained about 50 percent resiliency at 1400 °F and no resiliency at 2000 °F. Higher temperature materials will be required for future seal applications at 2000+ °F.
2. By themselves, the engineered core seal designs did not show improvement in resiliency when compared to the baseline spring tube design. This is not surprising, as these seals did not possess an internal spring element. At high temperatures, these seal candidates also showed a drop in resiliency performance with values similar to the baseline spring tube seal design.
3. The use of canted coil spring preloader devices behind the seals yielded substantial improvements (up to approximately 3x) in resiliency when compared to the seal candidates by themselves. Addition of the preloaders also resulted in up to a 26 percent improvement in resiliency versus the baseline spring tube seal at room temperature.
4. The unit load values for the engineered core seals may be acceptable for CMC control surface applications, but exceeded the design limits for Shuttle tile TPS applications. Therefore in these situations, the seals must be used in conjunction with a less stiff preloading device. Room temperature testing on seals with a “medium-duty” canted coil preloader behind them demonstrated significant reductions in load (down to approximately 6 lbf/in.) when compared with the seals themselves.
5. Flow testing on the AC1 and TC1 engineered core alternatives showed a decrease in leakage rates when compared to the baseline seal design. Evaluations with a canted coil spring preloader behind the seals demonstrated an increase in flow rate relative to the seals by themselves. Due to the less stiff spring preloader, the seals were not compressed and did not fill the groove as well as the seals by themselves. The authors believe that further optimization of the seal/preloader/groove system should result in lower flow and improved seal resiliency as compared to the baseline design.
6. Preliminary compatibility testing conducted between various candidate seal materials (e.g. Nextel 440, 720) and CMC control surfaces (e.g. C/SiC and C/C) showed bonding at 2600 °F. Stickage between the materials could result in seal damage and/or control surface panel damage and therefore requires further investigation.

FUTURE WORK

GRC’s plan for developing new advanced control surface seal designs focuses on making improvements to each of the three main components of this seal architecture: the overbraid, the spring tube (or other resilient element), and the core. These improvements are aimed at improving seal resiliency, decreasing seal leakage, and enhancing seal durability.

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